Supersonic Flow in Micro-Channel and Its Application to High Heat Flux Cooling

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論 文 内 容 要 約

This dissertation has focused on the heat transfer enhancement of the air cooling system by using the supersonic flow inside a micro-channel. The contents have been presented in the following six chapters.

Chapter 1:　General introduction and background

The demand on the electronics cooling and the cooling technologies with the related problems were introduced. Based on the problems, advantages and previous study of the supersonic flow in a microscale were introduced. Consequently, the motivation and objectives of this dissertation was mentioned: the proposition of the micro-nozzle geometry, the evaluation method of the temperature field of the supersonic flow inside a micro-channel, the considerations of the optimum nozzle and channel geometry, the analytically and experimentally evaluations of cooling performance of the proposed heat sink.

Chapter 2:　Heat transfer by the supersonic flow inside a micro-channel

In chapter 2, the fundamentals of supersonic flow in micro-scale were explained. To realize the high heat flux cooling by the micro-channel heat sink utilized the supersonic flow, the micro-nozzle geometry was determined by the quasi-one-dimensional theory. Then, the geometry of the bump nozzle was proposed and the feasibility of the supersonic flow in a micro-scale on heat transfer was indicated.

From the benchmark simulations, it was confirmed that the numerical method in this study was applicable to the compressible flow in a micro-scale despite the existence of the strong effect of viscosity. The superiority of the bump nozzle, compared to the Laval nozzle was shown by the comparison of the bulk mean temperature and the Nusselt number distributions, which were obtained from numerical simulations. The bump nozzle has advantages, such as the high manufacturability because of its simple geometry but can be form the low temperature flow due to the adiabatic expansion.

Chapter 3:　Evaluation of temperature field of supersonic flow inside a micro-channel

In chapter 3, to confirm the applicability of visualization and evaluation of the density field of supersonic air flow inside a micro-channel using an interferometer, preliminary experiment and numerical simulations were performed. The micro-channels were constructed through a MEMS process. The channel depth, which is equal to the length of the optical path, was determined from the estimation of the Gladstone-Dale relation. Here, the trade-off between the channel depth and expected fringe number was considered.

The measurement was conducted using a phase-shifting interferometer. The phase-shifting technique allows for high accuracy measurements because the phase difference between each interferogram, which has different polarized angles, is known. The refractive index distributions were measured as brightness distributions. The density distribution was evaluated from the measured brightness distribution and compared with the calculated distribution. Then, the numerically obtained density distribution showed the good agreement with that of experimental result, as shown in Fig. 1. Hence, it was indicated that the numerical data of the temperature distribution can be used for the cooling performance evaluations.

The measured density distribution of the supersonic flow inside the bumped micro-channel was the roughly the same distribution with that of the numerical data, as shown in Fig. 1. Therefore, the generation of the low temperature flow was indicated. To explain the difference of the density distributions between the numerical and experimental data, the surface roughness was measured by using the shape measurement laser microscope. The arithmetic mean roughness on the wall surface of 11.28 μm, which was the 4.8% of the designed value of the channel width, was obtained. It showed that the surface roughness was not a negligible matter.

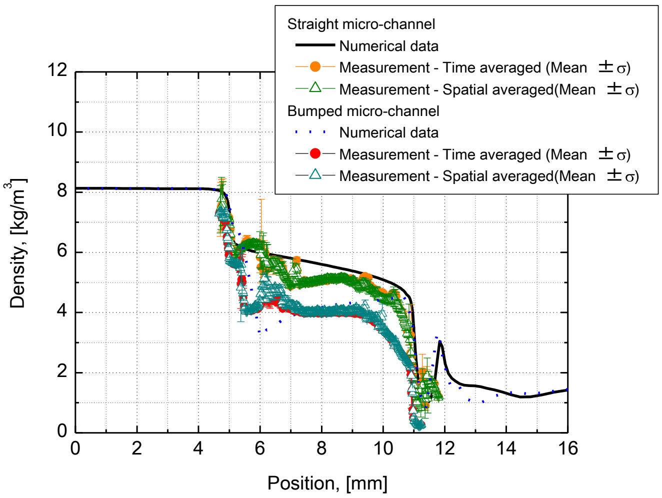

Fig. 1 Comparison of the density distributions between the straight and the bumped channel.

Chapter 4: Considerations of the optimum nozzle width and channel geometry

In chapter 4, to investigate the optimum geometry of the micro-channel as the flow passages in the micro-channel heat sink, the effect of the micro-channel shape on the cooling performance was evaluated. The geometry of the bump nozzle was consisted of simple arc curvature. This kind of simple geometry with high performance is one promising candidate for the geometry of the flow path of the heat sink. However, the bump nozzle has a fatal flaw in the geometry on the manufacturing process. The bump nozzle as the C-D nozzle is the most important part to generate the supersonic low temperature flow, but it is precisely required to set the each bumps at the completely the same position because of its scale of micron. Thus, the unilateral bumped micro-channel was proposed to improve the feasibility of the micro-channel heat sink utilized the supersonic flow. In addition, the diverging shape was applied to the micro-channel, which was placed at the downstream of the nozzle section.

From the numerical simulations, the flow field of the proposed nozzle flows was analyzed. By using the numerical results, the effect of the existence of unilateral bump nozzle and nozzle width were discussed. Figure 2 shows the distributions of the changes of bulk mean temperature and the heat transfer coefficient h on the walls. The h distributions have a difference at the nozzle section due to the existence of the unilateral bump nozzle. The values of h were high at the nozzle region because the thermal boundary layer was not well developed. At the downstream of the nozzle section, the h was almost constant around 500 W/(m^2·K). The micro-scale effect was contributed to keep h such a high value comparing to the values of forced convection of gas. In addition, this value of h is almost same as that of the natural convection of liquid and the part of the forced convection of the liquid. From these things, it was shown that the unilateral bumped micro-channel is a great candidate to the geometry of the heat sink for the electronics cooling.

The downscale of the channel size, the heat transfer area is increased. However, the pressure loss is increased with the decrease of the channel width. Thus, the maximum cooling performance is determined by the trade-off between the number of channels and the flow field. Therefore, the relation between the channel width and flow temperature were also investigated, as shown in Fig. 3. Then, it was shown that the nozzle width of 200 μm with the diverging micro-channel was the optimum geometry for the high heat flux micro-channel heat sink.

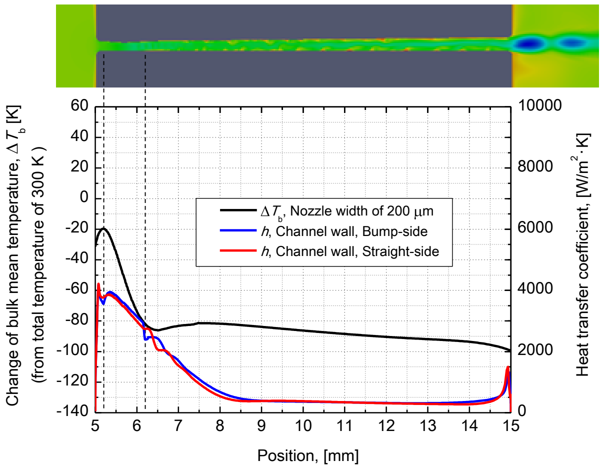

Fig. 2 Evaluated bulk mean temperature and heat transfer coefficient distributions of the unilateral bumped micro-channel.

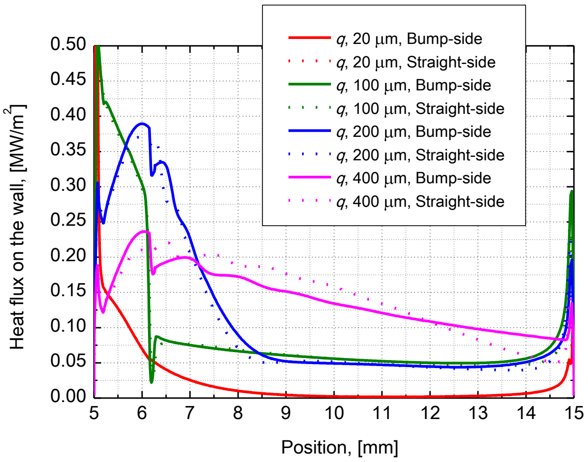

Fig. 3 Comparison of the heat flux on the walls, which were Bump-side and Straight-side of the micro-channel,

Chapter 5: Evaluations of micro-heat sink

In chapter 5, to validate the feasibility of the micro-channel heat sink, which includes the unilateral micro-nozzle and diverging downstream, cooling performance evaluations were analytically performed. Then, it was shown that the proposed micro-channel heat sink with the fin thickness of 200 μm and the fin height of 15 mm has approximately 1.5 MW/m^2 of cooling performance. Then, the number of channels was determined to 22.

To experimentally evaluate the cooling performance, proposed micro-channel heat sink was fabricated. The details of micro-channel heat sink and experimental set up are shown in Fig. 4. The air compressed to 0.7 MPa under the room temperature passed through the micro-channel heat sink to the atmosphere of 0.1 MPa, which are the same as the boundary condition on numerical simulations. To realize the performance test experimentally with the high heat flux more than 1.0 MW/m^2 and to measure the cooling performance as the heat flux, heat concentrator was designed. The heat concentrator was made of copper and has the halls to put the thermocouples (TCs). The distance between each hall was known. Then, the heat flux in +*z* direction can be determined by the measurement of the temperature distributions. The top surface of the heat concentrator simulated the surface of the heated electronics. In addition, the electronics have the maximum temperature to save their architecture and performance. Thus, the heat flux was generated from the heater and the output of the heater was controlled by temperature controller with the PID system to keep the surface temperature of 80°C at the top of the heat concentrator. To evaluate the method of the heat flux measurement, the heat conduction analysis was conducted by using OpenFOAM.

As shown in Fig. 5, the temperatures inside the converging section of the heat concentrator were changed in the range of 10 K, but the temperature gradients were almost constant during the measurement. Thus, the heat flux can be evaluated from these temperature gradients. The difference of the temperature gradient due to the selected measurement points was observed. The temperature gradients from the measurement points of TC5 and TC7 were obtained at the range of 1.0–2.0 MW/m^2, despite the temperature gradients from TC5–TC6 were lower than that of 0.5 MW/m^2. It was caused by the effect of the non-uniform temperature distributions of the *x*-*y* plane at the position of *z*=0–100 mm. Therefore, the micro-channel heat sink utilized the unilateral bump nozzle and the diverging downstream geometry has the cooling performance of around 1.0 MW/m^2 despite the air cooling device, even the thermal insulation was not perfect.

After the measurement, the torn off fin plates were found. It might be caused by the supersonic flow oscillations and the thermal stress during the experiment. The shock waves, which occurred at the exit of the micro-channel, made the micro-channel walls broken because of its vibration related with the aerodynamic sound. Therefore, it was indicated that the optimum geometry of the micro-channel heat sink, which utilized the supersonic flow, has to be considered the strength of the fin plates.

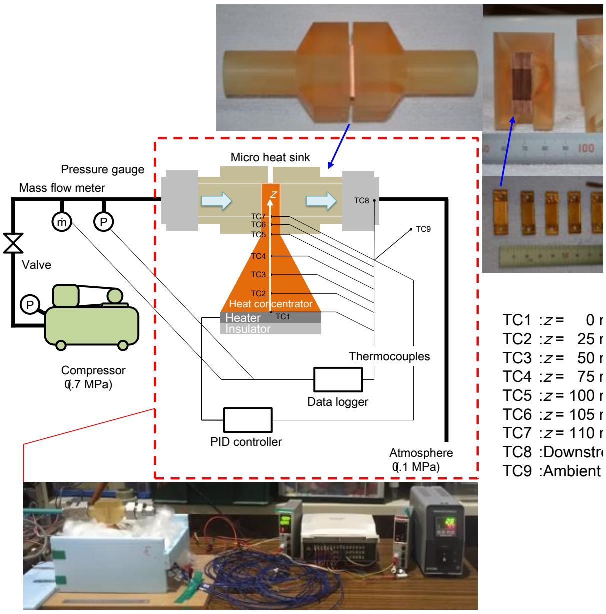

Fig. 4 Experimental setup for the cooling performance test. Right upper photograph shows the micro-channel heat sink (fin plate array) with the casing. Right lower photograph shows the each fin plates and the connection rod for the fin array. Downside photograph shows the appearance of the experimental setup.

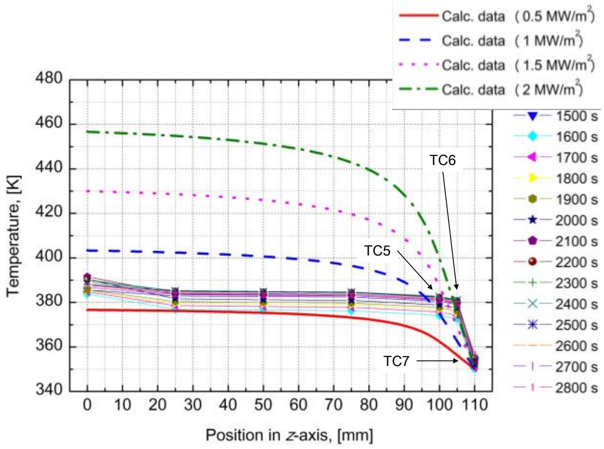

Fig. 5 Measured (1500–2800 s) and numerical simulated temperature distributions inside the heat concentrator.

Chapter 6: General conclusion

In chapter 6, the conclusions obtained from the each chapter were summarized, which will contribute to realize the micro-channel heat sink as the high heat flux cooling device. The obtained conclusions were as follows.

1. The numerically obtained T_b and Nu distributions were compared between the flow at the downstream of Laval nozzle and that of bumped nozzle. From this comparison, it was shown that the cooling performance of bump nozzle is approximately the same as that of Laval nozzle despite the shape is very simple. In addition, the maximum width of bump nozzle is narrower than that of Laval nozzle. Therefore, the number of channels, which indicates the heat transfer area, with bump nozzle is larger than that of Laval nozzle.
2. The density distributions of supersonic air flow inside the micro-channels were successfully measured by using phase-shifting interferometer, despite the channel depth of only 500 μm. From the comparison of the density distribution between the experimental and numerical data, it was shown that the temperature distribution can be evaluated from the numerical simulation.
3. The unilateral bump nozzle and the diverging micro-channel geometries were proposed to improve the feasibility of the high heat flux micro-channel heat sink. The optimum nozzle width of 200 μm was found by considering the effect of the nozzle width and the trade-off between the requirement of high heat flux and the number of channels.
4. From the experimental cooling performance test, it was evaluated that the micro-channel heat sink, which includes the unilateral bump nozzle and diverging downstream geometry, has the cooling performance of 1.0 MW/m^2 despite the air cooling device.